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ABSTRACT

The paper describes the results of the experimental study of the compressors (3 – 5 ton), motor of which have been cooled by the discharge gas. The applied method to cool of a compressor motor is based on the phenomenon known as the Ranque – Hilsch Vortex Tube that has no moving parts and separate incoming gas flow into hot and cold fraction. In the compressor high-pressure discharge gas enters the electric motor cavity tangentially through the inlet opening and develops axisymmetric vortex flow. The spinning stream of the gas revolves toward periphery of the cavity and exhausted partially as a hot flow fraction through a specially design opening while a cold fraction of the flow, still spinning, is forced to exit through the motor stator-rotor gap. This cold fraction of flow exhausted through the circumferential stator-rotor gap effectively cooled electric motor, thereby enhancing motor and overall compressor operating efficiency.

INTRODUCTION

Various arrangements have been provided for the purpose of cooling the compressor motor, among the more prominent of which are use of a gas stream passed into contact with hot motor parts, or introduction of vaporizable liquid (liquid refrigerant) into a hermetic motor casing and permitted to vaporize in contact with heated components.
The method of cooling compressor motor defined very much by type of gas distribution system which can be subdivided, mainly, as follows:
1. Low side configuration in which the housing cavity is at suction gas pressure.
2. High side configuration in which the housing cavity is at discharge gas pressure and the suction gas enters directly into the compressor suction chamber.
3. A partition configuration, which combines, separated suction gas and discharge gas pressure chambers inside of the housing cavity.
In the case of low side and partition configuration, gas of relatively low temperature is permitted to pass through the electric motor before being introduced into compression chamber. Very often low side configuration uses a semi-direct suction system in which a main portion of the refrigerant gas of relatively low temperature is drawn from the interior of the compressor housing directly into compression chamber, while another portion of the refrigerant gas is drawn through the motor rotor-stator gap to effect cooling of the motor therefore [1]. In the low side and partition configurations the suction gas overheated as by the operating electric motor so by a hot discharge system parts inside of the housing. Heat transfer to the suction gas in a hermetic compressor is known to have adverse effect on compressor performance [2]. Theoretical estimates show that a 10° F increase in suction gas temperature results in approximately a one percent decrease in compressor efficiency.

In a high side compressors compressed gas discharged inside of the housing and heat transfer phenomena affect not only temperature of the suction gas, but also the temperature of the motor. The high side design uses direct suction, thereby minimizing the suction gas heating by the discharge system parts. Because the motor operating efficiency decreases as the motor temperature increases due to heat absorbed from the surrounding discharge gas, the overall compressor efficiency is adversely affected. Tests show also that for every 10° F increase in motor operating temperature, insulation life of the motor winding is cut in half.

In overcoming the above described disadvantages related to motor cooling of high side configuration compressors, there is provided method to cool motor of the compressor by using the phenomenon found by G.J. Ranque and disclosed in U.S. Patent 1,952,281 [3].

THE VORTEX PHENOMENON

Ranque referred to vortex tube gas streams as counter-flow and uniflow types although it has since appeared that the counter-flow type is superior for the emission of separate cold and hot gas streams [4, 5]. A counter-flow vortex tube shown in Fig.1 has diaphragm with hole on cold gas output end, one or more tangential gas input nozzles with spinning chamber for producing vortex, and throttling valve on the hot gas output end. The exits from the spinning chamber will extend endwise and preferably parallel to the axis, but will have different areas and extend to different distances from the swirl chamber. The gas set in a vortex motion revolved toward the hot end and swirl will build up pressure in the outside perimeter due to centrifugal force, which will exceed the pressure at the axis of the swirl. The pressure **P** at the periphery of the swirl is related to the pressure $\mathbf{P_0}$ at the axis as follows:

$$\mathbf{P = P_0 \exp (M\, \omega_S^{\ 2}\ R_S^{\ 2} / \ 2RT_i)} \qquad \mathbf{(1)}$$

Where **M** is the molecular weight of the gas; **R** is the gas constant; $\mathbf{T_i}$ is inlet gas temperature; $\mathbf{R_S}$ **is** the radius of the spinning chamber; $\mathbf{\omega_S / 2\pi}$ **frequency** in revolution per sec.; $\mathbf{\omega_S}$ is the peripheral velocity. C.D Fulton [6] determined that the temperature of the three streams of gas are related by the following improved energy balance:

$$\mathbf{f\,(Ti\ -\ T_C - JT) = (\ T_H - T_i\ - JT)} \qquad \mathbf{(2)}$$

$\mathbf{T_C}$ is temperature of cold flow fraction; $\mathbf{T_H}$ is temperature of hot flow fraction, **f** is cold fraction = mass flow of cold gas/ mass flow of inlet gas; **JT** is Joule – Thomson temperature drop of gas on adiabatic throttling from inlet state to outlet pressure.

According to Fulton, fifteen important parameters have to be considered in process of the vortex effect devices design and problem of optimization my be performing only by a very large number of parametric experiments, where parameters are changed step by step with following measurement of pressure, temperature, mass flow, etc. There is no agreement at present time as to exact theory of operation of Ranque- Hilsch vortex tube.

Optimization of the vortex tube performance brought equation based on the dynamic heat exchange theory:

$$\mathbf{\Delta T = [\varphi(1- f)/ 1+ \varphi(1-f)]\ \{1 - [\ 1 + (1 +2\alpha\beta\, f^2\, \gamma^{\ 2}\ )\ \exp 0.5\ /\ \ 2\, \gamma^{\ \delta}\ ]\ \}} \qquad \mathbf{(\ 3\ )}$$

where $\boldsymbol{\varphi}$ =(A/B)St **=** 1.95 is optimization parameter; A is surface area available for heat transfer; B is cross-sectional area of the stream; St is the Stanton number; $\boldsymbol{\gamma} \mathbf{= P/ P_0}$ is pressure ratio through the vortex tube; $\mathbf{f = F_I / F_D}$; $\mathbf{F_I}$ is inlet nozzle area; $\mathbf{F_D}$ is diaphragm cross sectional area; $\boldsymbol{\alpha = (\alpha_t / \alpha_d)^2 T_H / T_i}$; $\boldsymbol{\alpha_t}$ and $\boldsymbol{\alpha_d}$ are the orifice coefficients for the inlet nozzle and for diaphragm; $\boldsymbol{\beta} = (K-1) (2/K + 1)^{(K+ 1/K-1)}$ is thermodynamic constant; K is adiabatic exponent of the gas; $\boldsymbol{\delta} = K – 1/K$.

Numerous interpretations shown in the literature for the temperature difference produced by the vortex effect, but common explanation of axial stream cooling have been explained as adiabatic expansion process and heating of the peripheral stream related to centrifugal pressure.

There is a difference of opinions in the explanation of energy transfer from the axial stream to the peripheral one. Such phenomena generally attributed to the exchange of mechanical work caused by the velocity difference between the peripheral and core gas streams [7]. Kurosaka has attributed the Ranque-Hilsch phenomenon to the whistle sound, which produced by vortex flow [8]. Departing radically from disclosed theories Kotelnikov demonstrated experimentally that vortex phenomenon is based on acoustic standing wave generation process [9]. The difference of temperature between hot and cold end of Ranque-Hilsch vortex tube has been formulated as follows:

$$\mathbf{\Delta T = 2A \cos (2 \pi x / D_0 + \pi) \sin (\omega_S / C_0) \, e \exp (-B)} \qquad \mathbf{(4)}$$

where $\mathbf{x}$ is radial distance; $\mathbf{D_0}$ is tube diameter; $\mathbf{C_0}$ is speed of sound; **A** and **B** are constants. Various design modifications of the Ranque-Hilsch vortex tube have been discussed in the literature. U.S. Patents [10,11] disclosed vortex cooler comprising a scroll with gradually tapered spiral duct, as well as a cold exit nozzle and shaped as frustum of cone an expansion chamber. While passing along scroll channel the gas stream get swirled and accelerated to a supersonic velocity, whereupon it is discharged from the scroll into expansion chamber. While passing along a helical pathway over the chamber walls the gas stream reaches the blank cover, after which part of the stream is reflected from it and returns backward along the axial line to be discharged as a cold gas portion while hot portion of the gas discharged through the ports in the blank cover.

A vortex tube can produce temperatures from - 40°F (- 40°C) to 230° F (+ 110° C) and can hold output temperature ±1° F (± 0.6°C). There are instances where the hot gas of the vortex tube is usable for heating and others where the hot and cold fractions are usable alternatively.

COMPRESSOR MOTOR COOLING SYSTEM

The sectional side elevation of the hermetic reciprocating high side compressor driven by refrigerant cooled motor and utilizing vortex tube phenomena to improve motor cooling is shown in Fig.2 [12]. The gas after compression discharged into the plenum 1 in the cylinder head, which is in fluid communication with cavity 2 formed in the compressor crankcase 3 and used to accommodate electric motor 4 winding end 5. The end wall 7-support stator 8, which is affixed to the wall 7 edges 9 by a plurality of, bolts 10. The hole 6 disposed at an angle to the circumferential end wall 7 and helps introduce the discharge gas or vapor into the crankcase cavity 2 (vortex spinning chamber) as a jet tangential to the cavity periphery. The discharge gas revolving in the vortex spinning chamber will be separated into hotter and colder fraction due to the fact that crankcase cavity 2 provide a generally cylindrical open interior the length of which being in any case small as compared to its diameter. It is important that swirl chamber have an open interior so that the swirl can follow the general contour of the periphery substantially around the axis of the spin chamber without formation of eddies which will dissipate kinetic energy as heat.

First circumferential gap 11 (see Fig. 3) used as the cold exhaust is provided between rotor 12 and stator 8 and is preferably approximately 0.030" wide. Second gap 13 used as the hot exhaust is circumferentially located between stator 8 and separating plate 14 and is preferably approximately 0.050" wide. The spinning discharge gas vortex moves in a direction away from the axis and toward inner part of the chamber 16. A definable portion of discharge gas is propelled through path 15 and exits through second gap 13 into discharge chamber 36.

The remaining discharge gas is forced back through a central part 16 of spinning vortex, so as to flow in a direction opposite outer flow path 15 of spinning vortex. Inner flow path 18 moves in a direction away from housing and toward crankshaft 34. Spinning vortex effectively cools the discharge gas flowing through inner flow path 18. This cooler fluid flows through path 19 and exhaust through gap 11 into discharge chamber 36. In this manner, motor 4 are effectively cooled by the cooler fraction of discharge flow, thereby enhancing motor operating efficiency and overall compressor operating efficiency.

Stator 8 is affixed to crankcase 3 by a plurality of bolts 10. Separating plate 14 is disposed intermediate stator 8 and crankcase 3 and is provided with a bolt-receiving hole 50. Spacing washers 51 spatially separate stator 8 from separating plate 14, thereby establishing intermediate space and gap 13 as shown in Fig.4.

The dimensions of gap 13 perform function of the throttling valve and may be altered by placing multiple or various width washers 51 intermediate stator 8 and separating plate 14. The width of circumferential gap 13 determines the temperature and flow rate of the discharge gas flowing through cooler gas flow path 19 and through rotor/stator gap 11. Enlarging gap 13 reduces the temperature and flow rate associated with the discharge gas flowing through cooler gas flow path 19 and gap 11. Reducing gap 13 increases the temperature and flow rate of the discharge gas flowing through cooler gas flow path 19 and gap 11. In this manner, the compressor of the present invention utilizes the vortex tube phenomena to effectively cool the motor windings and accelerate the evacuation of discharge gas from discharge plenum 1 of crankcase 3, resulting in enhanced operating efficiency. Further, due to the high velocity and increased volume of discharge gas flowing through rotor/stator annular gap 11, rotor 12 is effectively lifted so as to reduce the load on the lower part of the main bearing.

EXPERIMENTAL TESTS RESULTS

The hermetic high side compressors modified for vortex cooling of the motor and low side compressors with semi-direct suction have been chosen for experimental evaluation. The design capacities of the tested compressors with R-22 refrigerant were 35000 Btu/h, 38000 Btu/h, and 58000 Btu/h. As described above, throttling of the peripheral (hot) discharge flow portion have been performed by change of the total thru -flow area of the gap 148 resulted in variation of the temperature in the axial (cold) fraction of the flow, which affects adversely the effectiveness of the motor cooling process. In the preferred embodiment, gap 148 has been optimized to obtain maximum cooling efficiency for the 35000 Btu/h capacity

compressors only and the optimum width of the gap 148 (0.050”) have been used also for larger capacity compressors without any parametric evaluation.
The thermocouples have been installed in the discharge plenum 1 (see Fig.2) and on the wall 30 of the hub 31 to record temperature of the cold and hot flow fractions. Performance of the compressors has been tested at ARI (45 ° /130 ° /65 ° F) and CHEER (45 ° / 100 ° / 65 ° F) operating conditions. The tests results are shown in table below and in Figures 5, 6, and 7.

Design Capacity, Btu/h	**Test condi-tion.**	**Temperature, ° F**				**Capacity, Btu/h**		**EER, Btu /Wh**	
		Dis-charge	**Axial**	**Oil**		**H.S.**	**L.S.**	**H.S.**	**L.S.**
				H.S.	**L.S.**				
35K,w/o Vortex	ARI	208	201	174	151	35083	35168	11.27	11.32
	CHEER	142	139	-	-	44320	43175	19.11	19.00
35K	ARI	208	179	160	151	35622	35168	11.57	11.32
	CHEER	142	127	-	-	46201	43175	19.83	19.00
38K	ARI	221	198	173	153	40403	38486	11.20	11.00
	CHEER	158	153	-	-	52776	48321	18.85	17.16
58K	ARI	203	190	171	161	63926	57979	10.88	10.66
	CHEER	161	152	-	-	79151	72496	18.34	17.32

H.S.-High side compressor; L.S.- Low side compressor

SUMMARY AND CONCLUSION

1. The proposed compressor motor cooling system is based on the phenomena known as the Ranque– Hilsch Vortex Tube, the device which has no moving parts and separate incoming gas flow on cold fraction exhausted from one end of the tube and hot fraction exhausted from another. The motor cooling system utilize the cavity under the motor as a spinning chamber, use motor stator-rotor gap for exhaust of the cool gas fraction, and define stator-crankcase gap for passage of the hot gas fraction.
2. Up to 30° F difference between discharge gas temperature and the temperature of the cold fraction passing through the motor rotor–stator gap have been recorded for the high side compressors driven by vortex cooled motor.
3. The increase in COP of the compressor utilizing the vortex tube phenomena to cool the motor stator winding is about 2%.
4. Developed motor cooling system enhances operating efficiency of the compressor and prolong life of the motor winding.
5. A thermodynamic formulation and optimization of the motor cooling system design parameters (pressure, temperature, rate of flow, geometric dimensions, etc.) required further experimentation and development.

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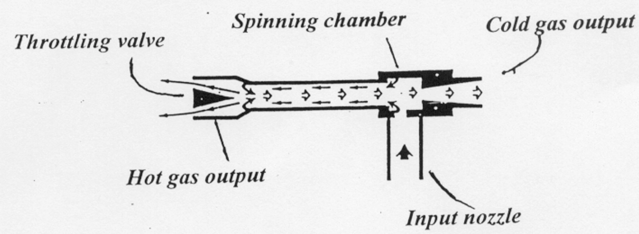

Fig.1. Ranque – Hilsch counter-flow Vortex Tube.

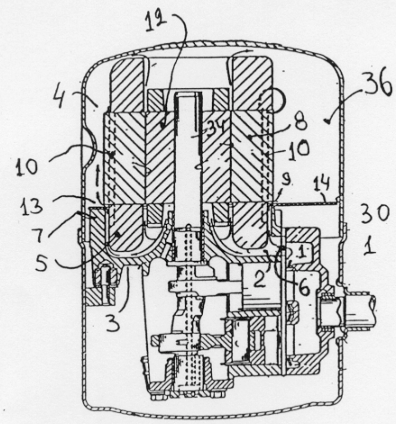

Fig.2. Hermetic compressor with vortex cooled motor

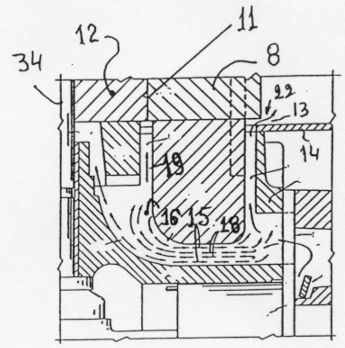

Fig.3. Hot and cold flow pattern of the discharge gas in the motor cavity

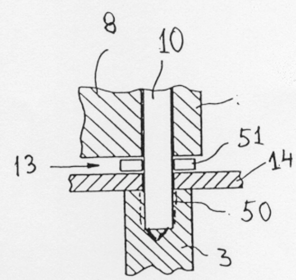

Fig.4 Assembly of the stator in the compressor comprising hot exhaust port

Fig.5 **TEMPERATURE OF THE GAS IN THE DISCHARGE CAVITY AND AT THE MOTOR INPUT SIDE (COLD FRACTION)**

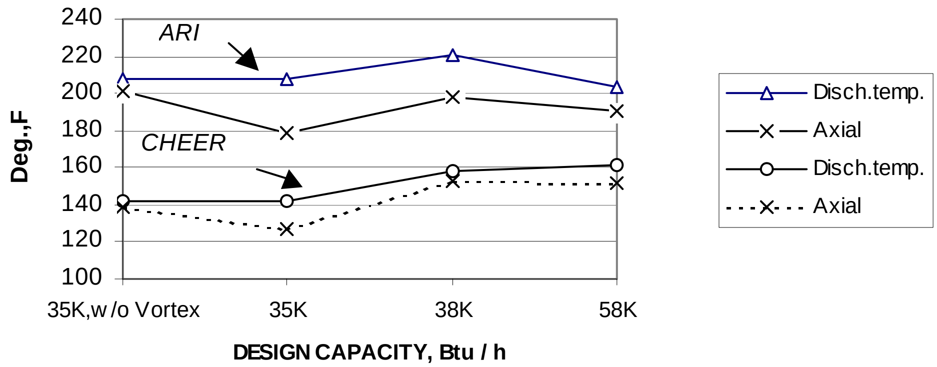

Fig.6 **CAPACITY OF HIGH SIDE COMPRESSORS WITH VORTEX COOLED MOTORS. ARI AND CHEER CONDITIONS**

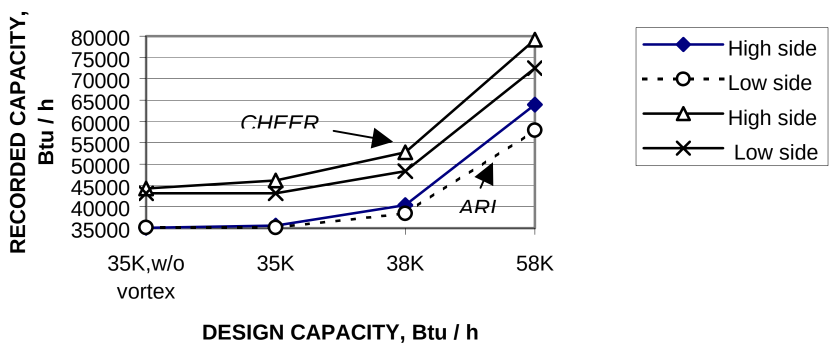

Fig.7 **EER OF HIGH SIDE COMPRESSORS WITH VORTEX COOLED MOTORS AT ARI AND CHEER CONDITIONS**